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Nakasaka

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(54) **IGNITION TIMING CONTROL SYSTEM FOR A SPARK IGNITION TYPE INTERNAL COMBUSTION ENGINE**

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F02P 5/152 (2006.01)
F02P 5/153 (2006.01)
F02D 13/02 (2006.01)
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F02D 15/04 (2006.01)
F01L 1/344 (2006.01)

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CPC ***F02P 5/145*** (2013.01); ***F02D 13/0238*** (2013.01); ***F02D 13/0269*** (2013.01); ***F02D 15/04*** (2013.01); ***F02D 35/028*** (2013.01); ***F02P 5/152*** (2013.01); ***F02P 5/153*** (2013.01); *F01L 1/3442* (2013.01); *F01L 2001/3443* (2013.01); *F02D 2200/021* (2013.01); *F02D 2200/0414* (2013.01); *F02D 2200/0611* (2013.01); *Y02T 10/142* (2013.01); *Y02T 10/46* (2013.01)

(58) **Field of Classification Search**
CPC F02D 35/027; F02D 35/028; F02D 41/00; F02P 5/145; F02P 5/152; F02P 5/153; Y02T 10/46
USPC 701/103–105, 110, 111; 123/406.11, 123/406.16, 406.19, 406.21, 406.25, 406.36
See application file for complete search history.

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(57) **ABSTRACT**

A base in-cylinder volume is corrected by a learned value to calculate a corrected in-cylinder volume. A difference of a crank angle which gives the corrected in-cylinder volume and the base crank angle is calculated as a "crank angle correction value", and a base ignition timing is corrected by the crank angle correction value and feedback correction value. An in-cylinder volume at a crank angle where a combustion ratio became a set ratio at the previous combustion is calculated as a "previous in-cylinder volume VFp", a difference ΔVFp of the previous in-cylinder volume VFp with respect to the base in-cylinder volume VBp at the previous combustion is calculated, and the learned value ΔVL is updated based on the in-cylinder volume difference VFp.

4 Claims, 14 Drawing Sheets

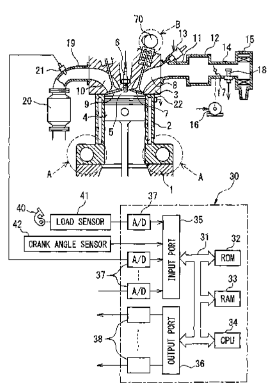


FIG. 1

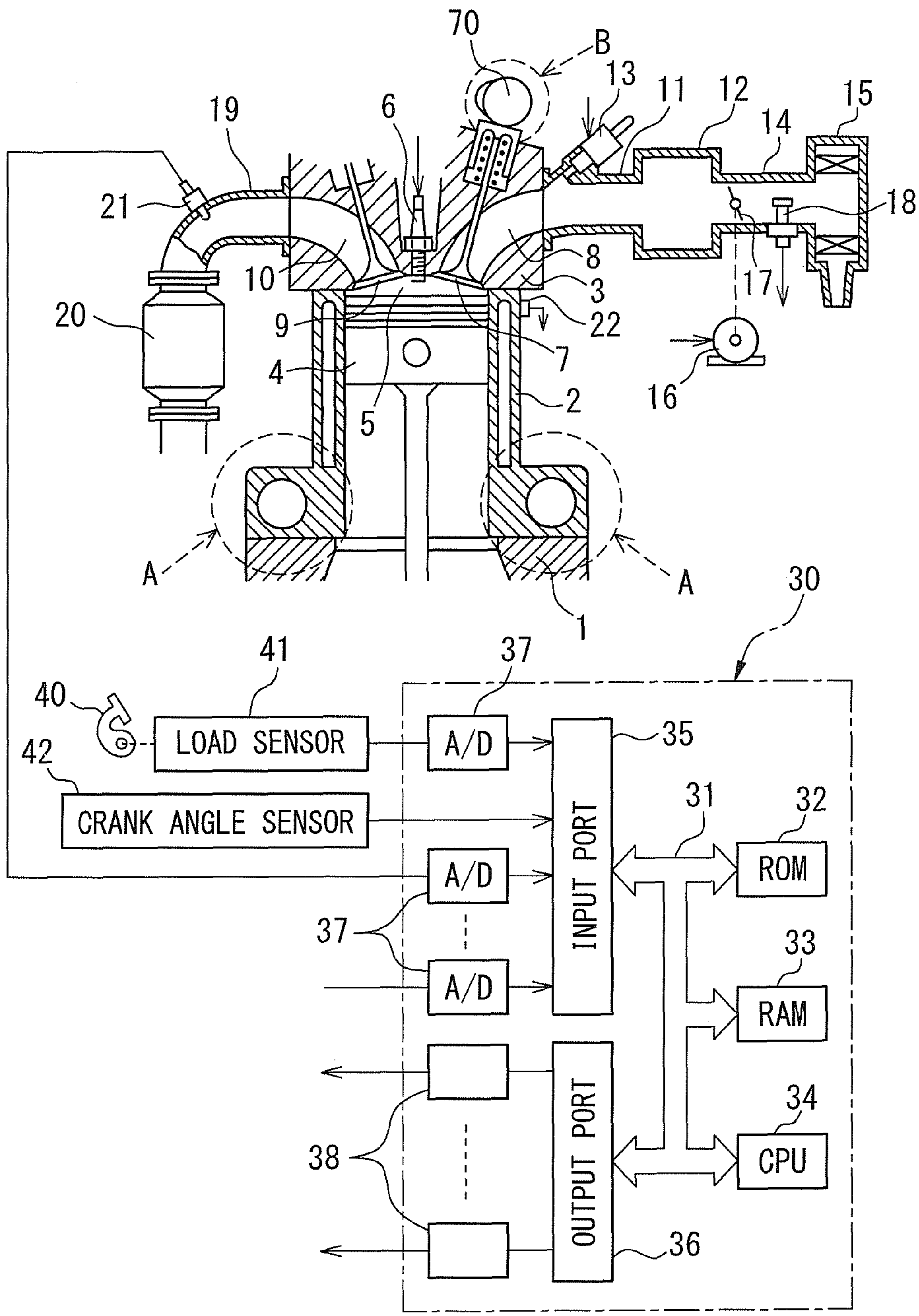

70
B
13
11
12
14
15
19
6
21
18
10
8
3
17
20
9
22
7
16
4
2
5
A
A
30
1
41
37
40
LOAD SENSOR
A/D
35
42
CRANK ANGLE SENSOR
31
32
A/D
ROM
INPUT PORT
37
33
A/D
RAM
OUTPUT PORT
34
38
CPU
36

FIG. 2

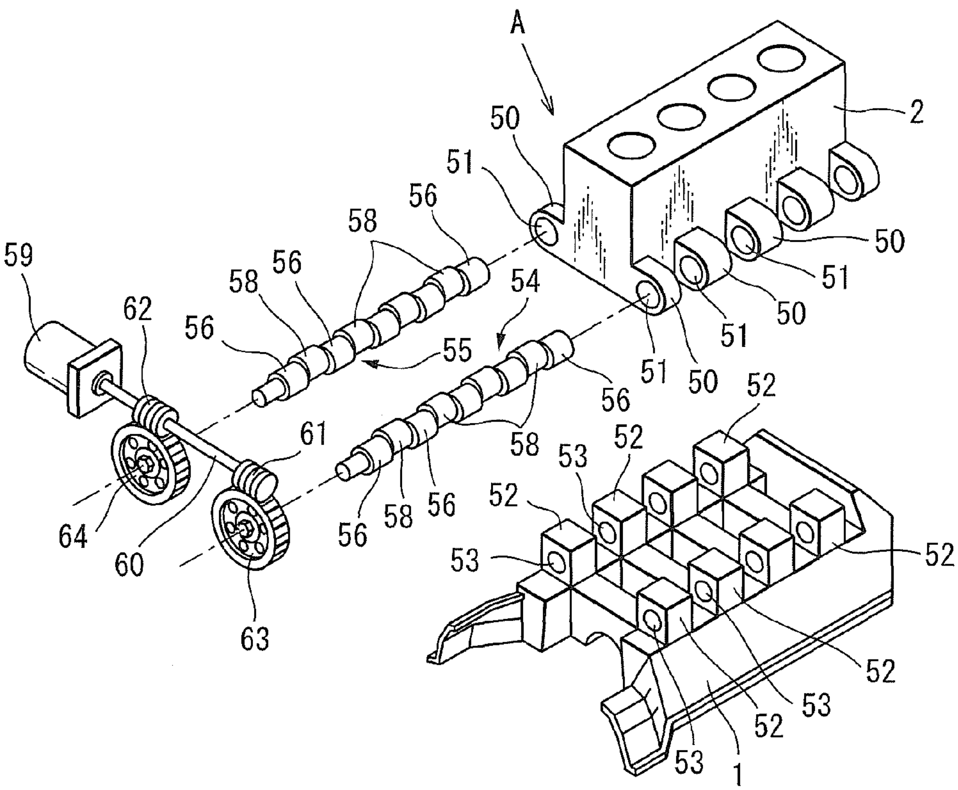


FIG. 3A

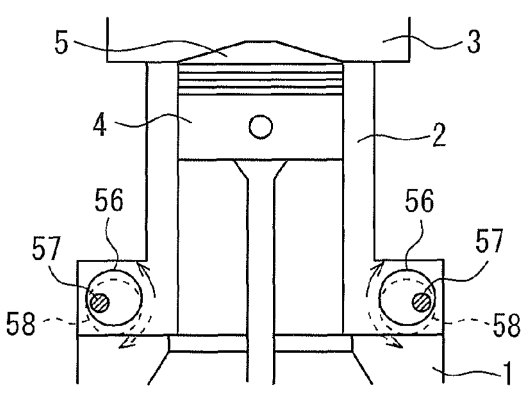


FIG. 3B

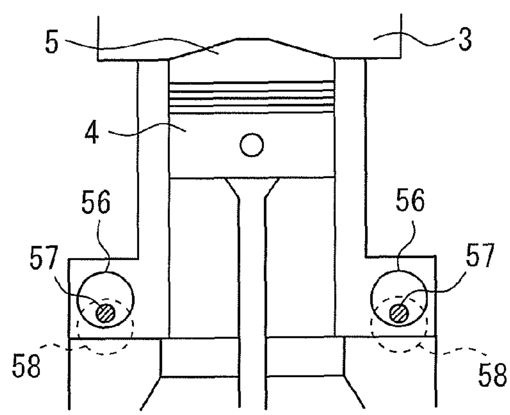


FIG. 4

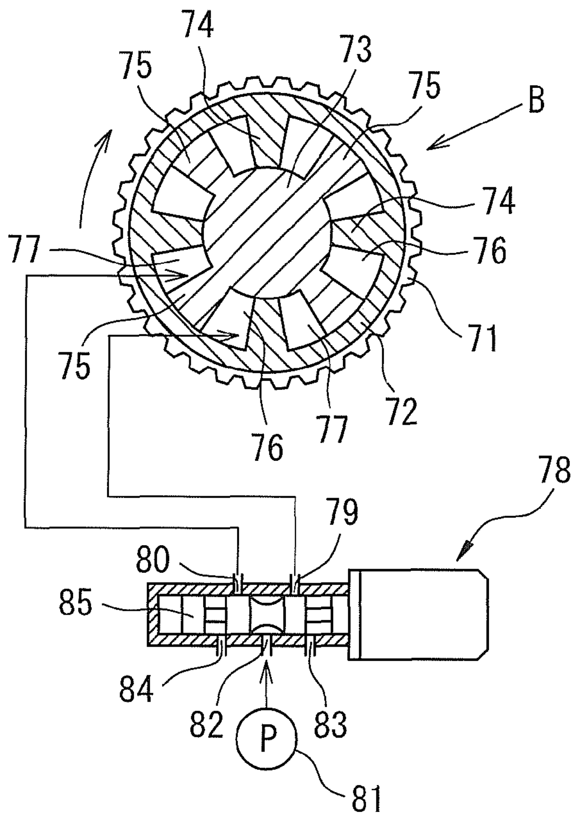

74
73
75
75
B
74
76
77
71
75
72
77
76
78
79
80
85
84
82
83
P
81

FIG. 5

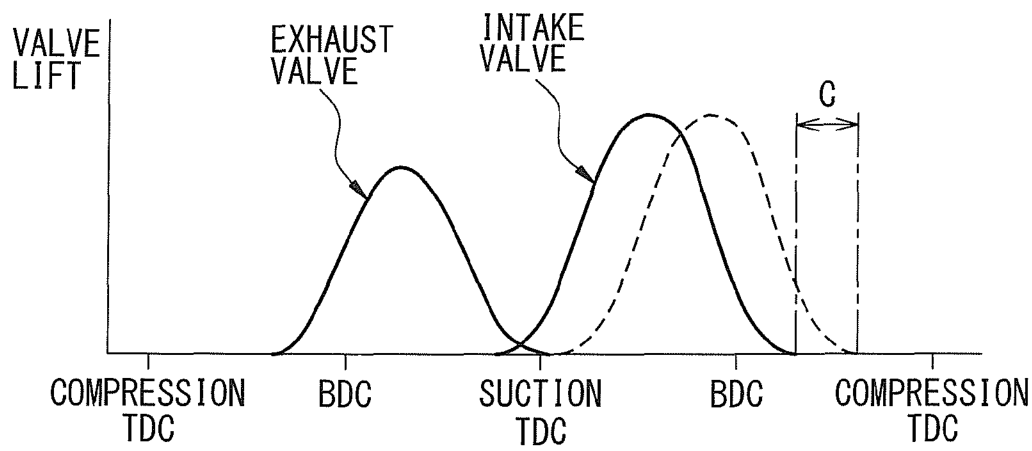

VALVE
LIFT
EXHAUST
VALVE
INTAKE
VALVE
C
COMPRESSION
TDC
BDC
SUCTION
TDC
BDC
COMPRESSION
TDC

FIG. 6

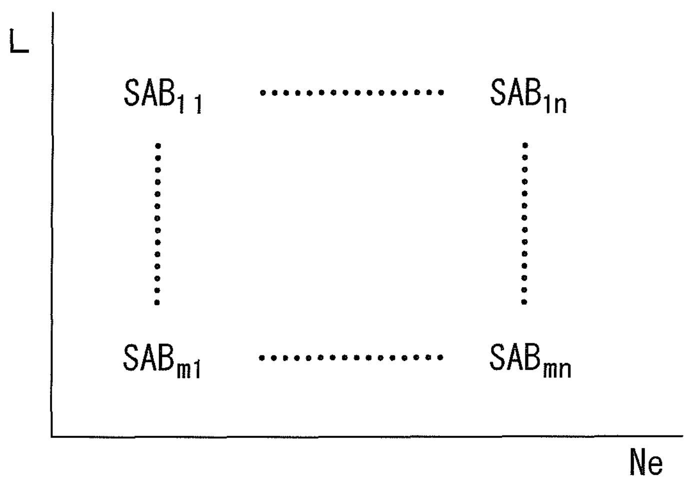

L
SAB11
SAB1n
SABm1
SABmn
Ne

FIG. 7

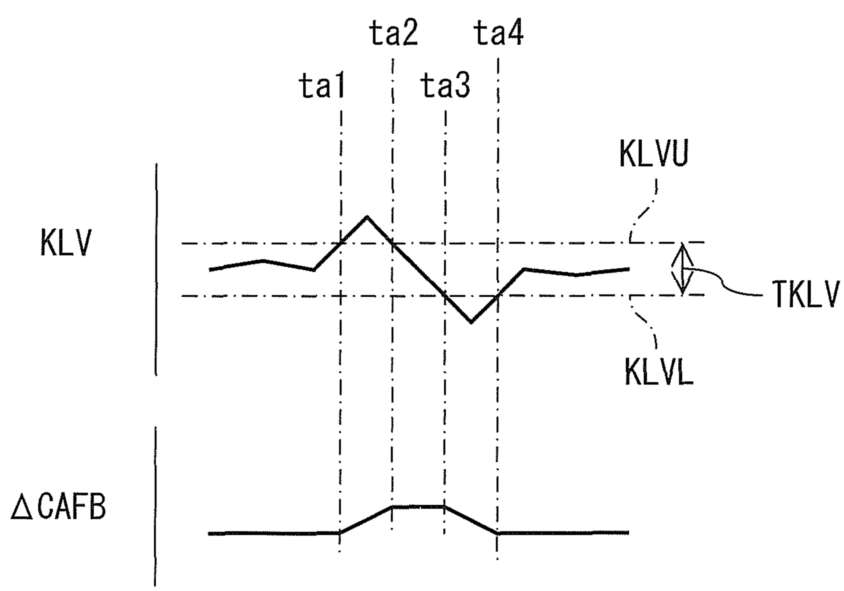

ta2
ta4
ta1
ta3
KLVU
KLV
TKLV
KLVL
ΔCAFB

FIG. 8

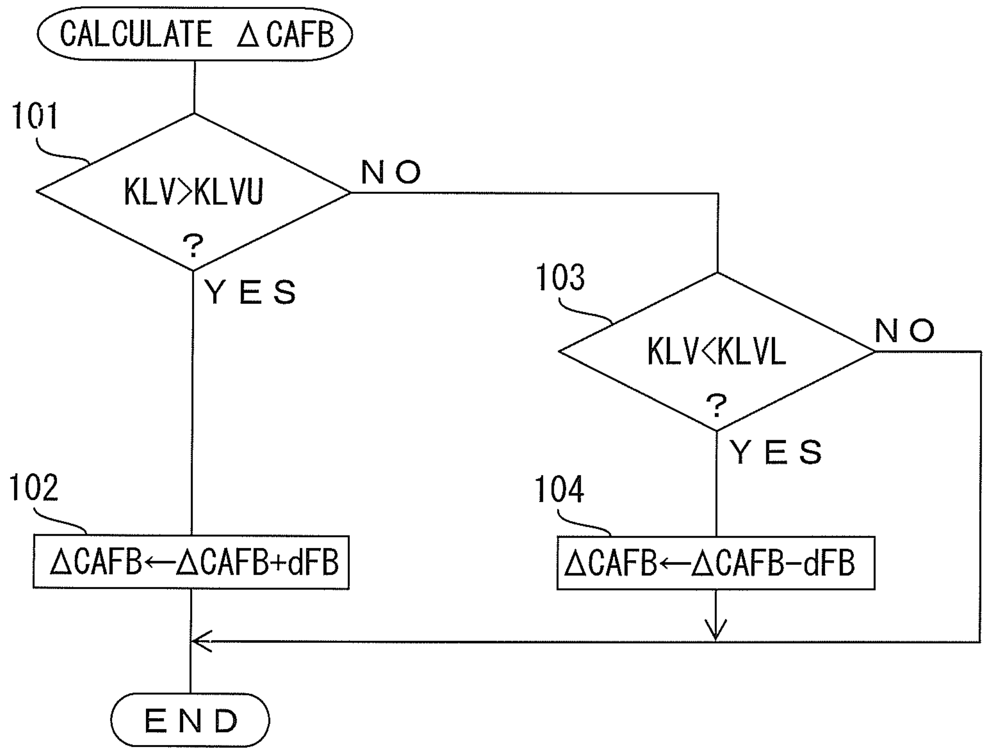

CALCULATE ΔCAFB
101
KLV>KLVU ?
NO
YES
103
KLV<KLVL ?
NO
YES
102
ΔCAFB←ΔCAFB+dFB
104
ΔCAFB←ΔCAFB−dFB
END

FIG. 9

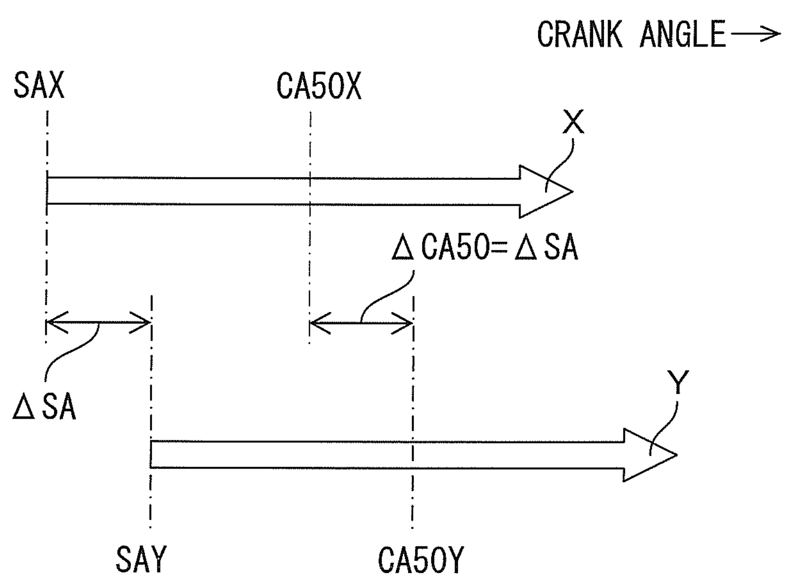

CRANK ANGLE→
SAX
CA50X
X
ΔCA50=ΔSA
ΔSA
Y
SAY
CA50Y

FIG. 10

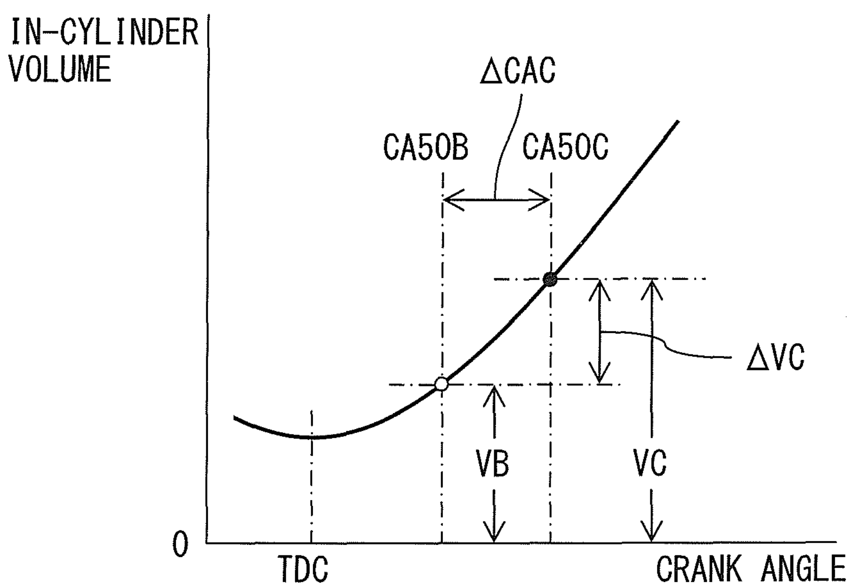

IN-CYLINDER VOLUME
ΔCAC
CA50B
CA50C
ΔVC
VB
VC
0
TDC
CRANK ANGLE

FIG. 11

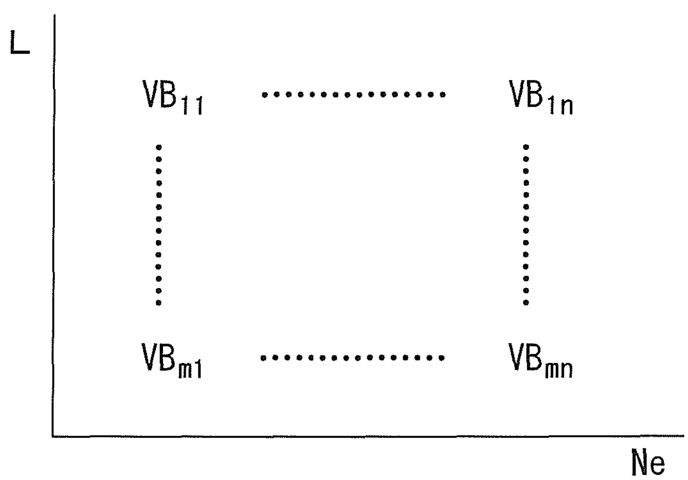

L
VB_{11}
VB_{1n}
VB_{m1}
VB_{mn}
Ne

FIG. 12

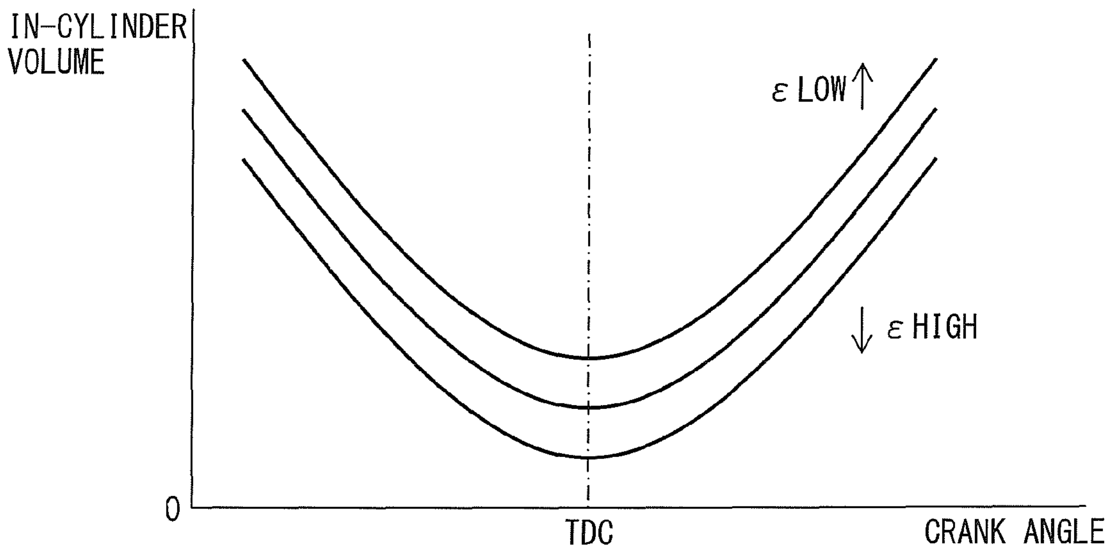

IN-CYLINDER VOLUME
ε LOW
ε HIGH
0
TDC
CRANK ANGLE

FIG. 13

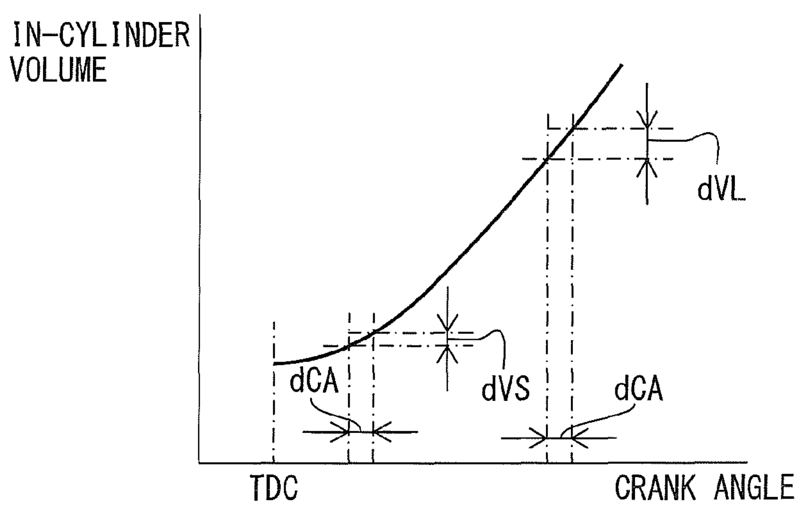

IN-CYLINDER VOLUME
dVL
dVS
dCA
dCA
TDC
CRANK ANGLE

FIG. 14

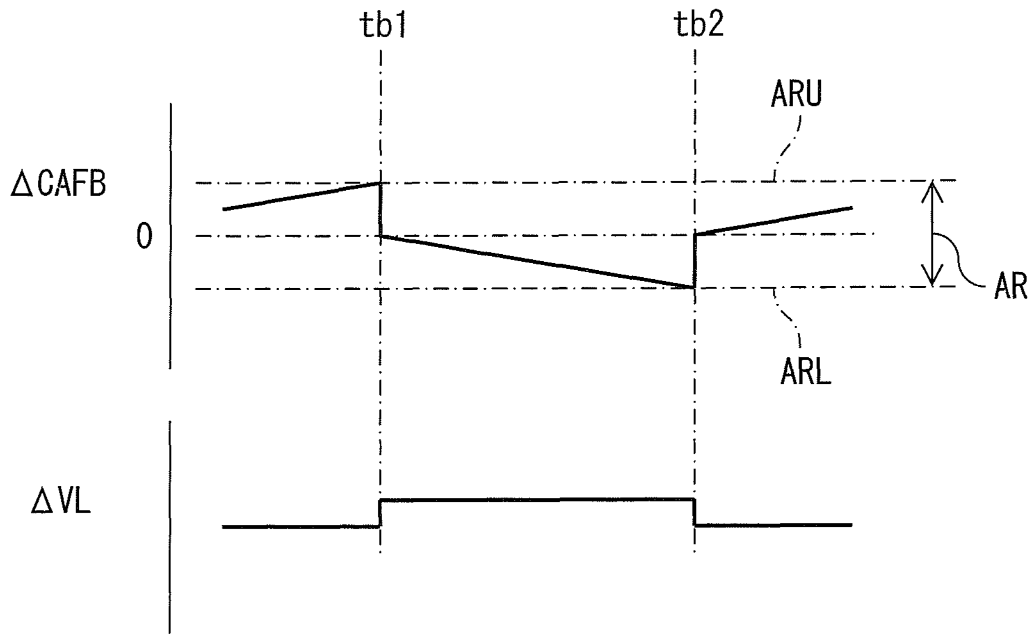

tb1
tb2
ARU
ΔCAFB
0
AR
ARL
ΔVL

FIG. 15

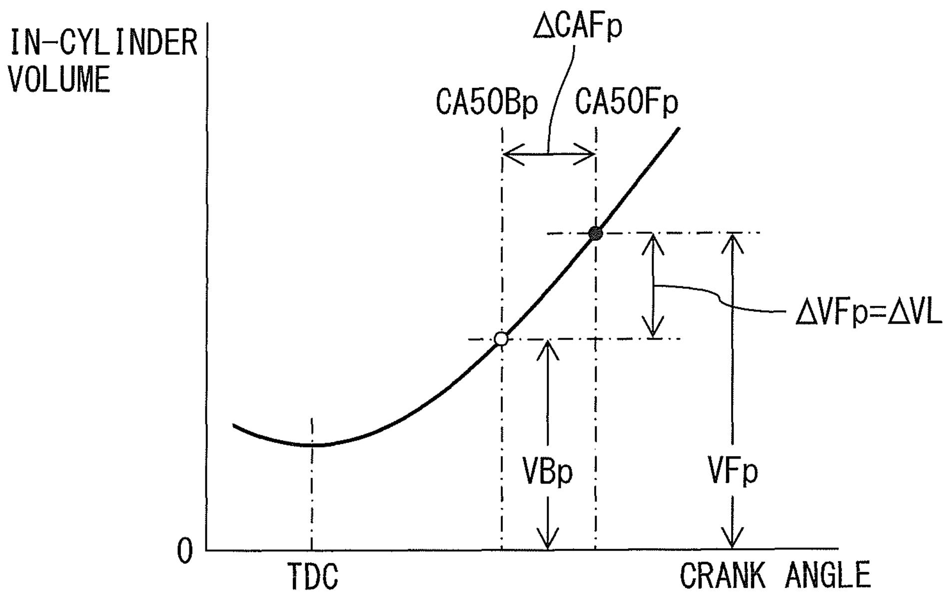

IN-CYLINDER VOLUME
ΔCAFp
CA50Bp
CA50Fp
ΔVFp=ΔVL
VBp
VFp
0
TDC
CRANK ANGLE

FIG. 16A

Ta	Ta_1	....	Ta_n
Δvd1	$\Delta vd1_1$	....	$\Delta vd1_n$

FIG. 16B

THW	THW_1	....	THW_n
Δvd2	$\Delta vd2_1$	....	$\Delta vd2_n$

FIG. 17

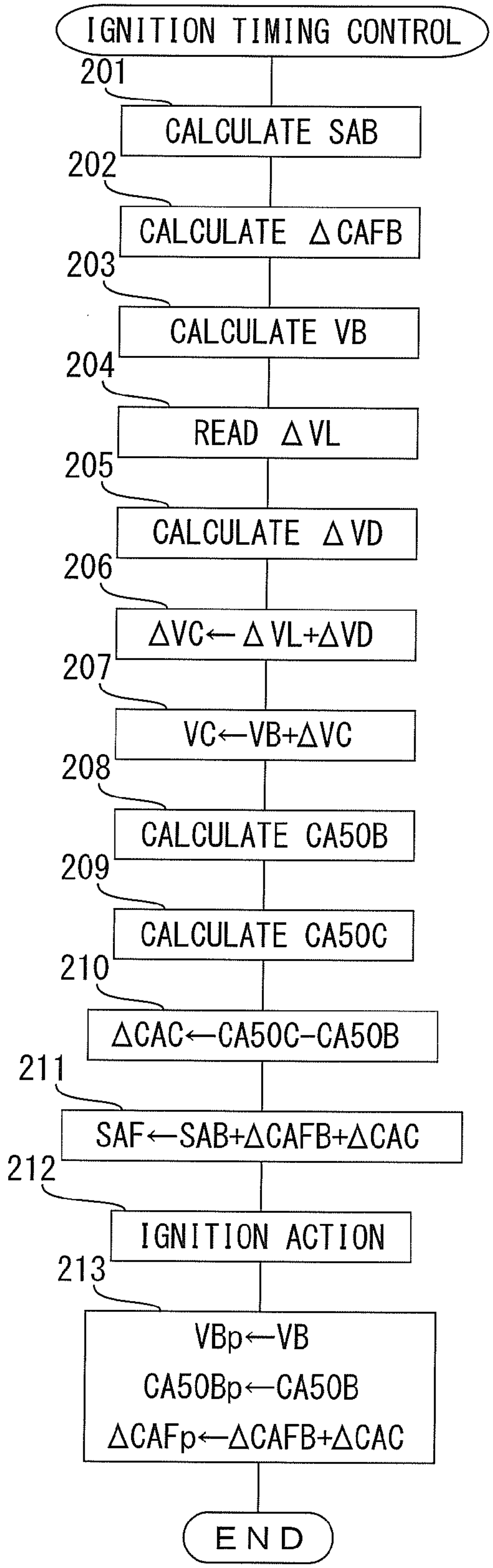

IGNITION TIMING CONTROL
201
CALCULATE SAB
202
CALCULATE ΔCAFB
203
CALCULATE VB
204
READ ΔVL
205
CALCULATE ΔVD
206
ΔVC←ΔVL+ΔVD
207
VC←VB+ΔVC
208
CALCULATE CA50B
209
CALCULATE CA50C
210
ΔCAC←CA50C−CA50B
211
SAF←SAB+ΔCAFB+ΔCAC
212
IGNITION ACTION
213
VBp←VB
CA50Bp←CA50B
ΔCAFp←ΔCAFB+ΔCAC
END

FIG. 18

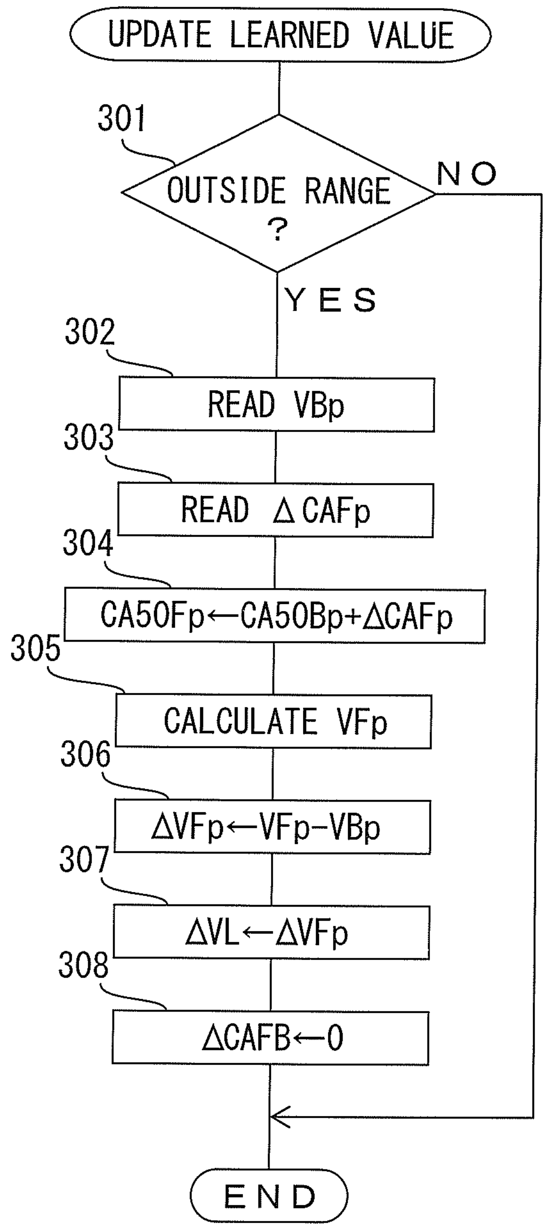

UPDATE LEARNED VALUE
301
OUTSIDE RANGE ?
NO
YES
302
READ VBp
303
READ ΔCAFp
304
CA50Fp←CA50Bp+ΔCAFp
305
CALCULATE VFp
306
ΔVFp←VFp−VBp
307
ΔVL←ΔVFp
308
ΔCAFB←0
END

FIG. 19A

Ta	Ta_1	....	Ta_n
rd1	$rd1_1$	....	$rd1_n$

FIG. 19B

THW	THW_1	....	THW_n
rd2	$rd2_1$	....	$Rd2_n$

FIG. 20

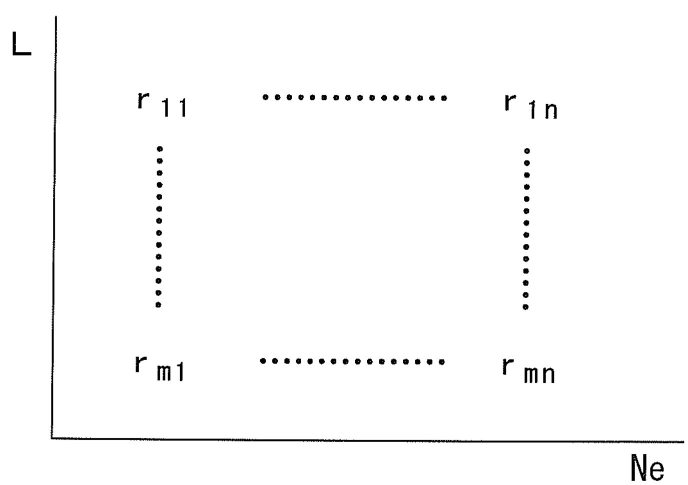


FIG. 21

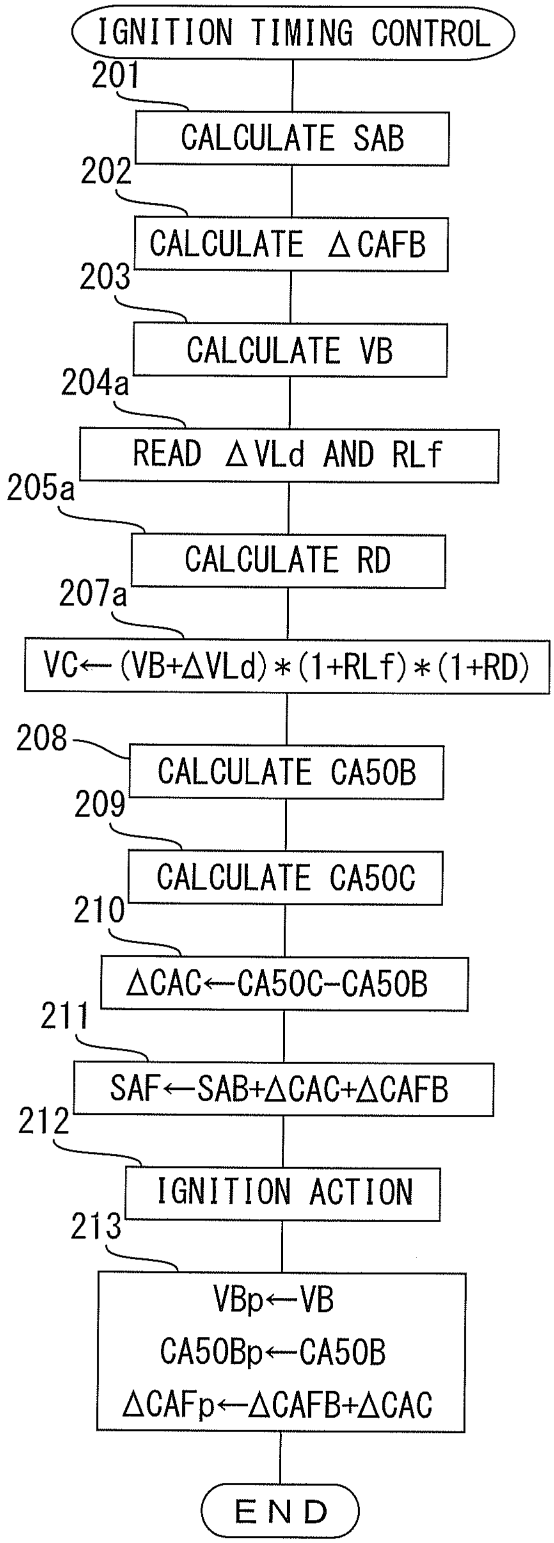
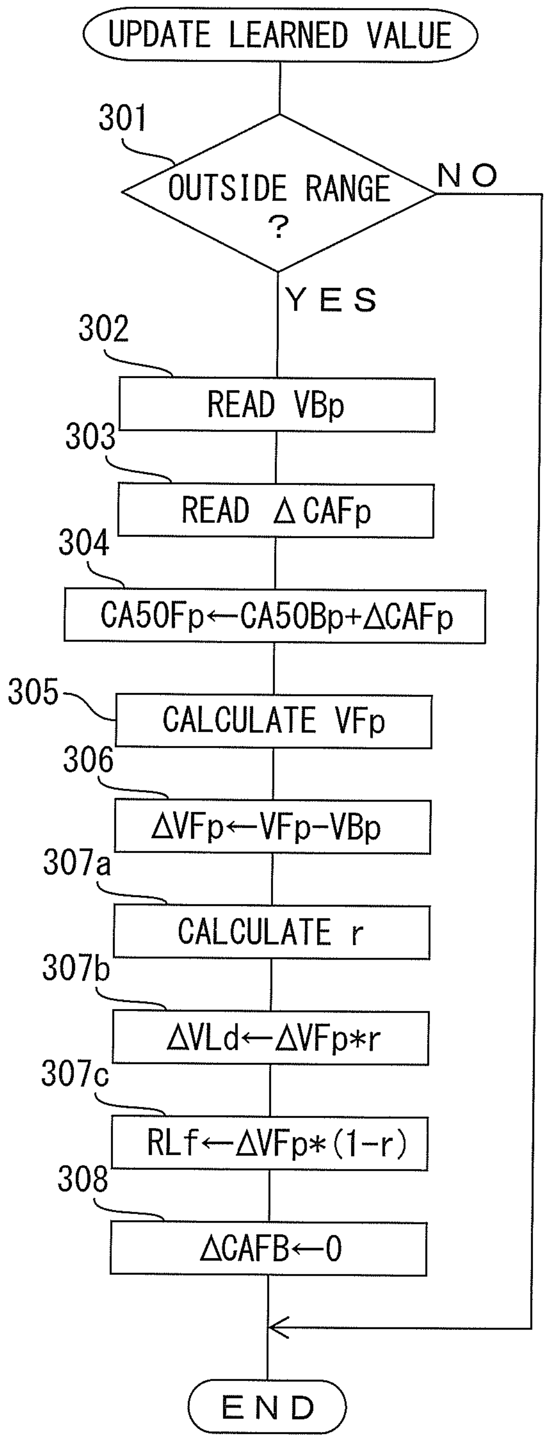

IGNITION TIMING CONTROL
201
CALCULATE SAB
202
CALCULATE ΔCAFB
203
CALCULATE VB
204a
READ ΔVLd AND RLf
205a
CALCULATE RD
207a
VC←(VB+ΔVLd)*(1+RLf)*(1+RD)
208
CALCULATE CA50B
209
CALCULATE CA50C
210
ΔCAC←CA50C−CA50B
211
SAF←SAB+ΔCAC+ΔCAFB
212
IGNITION ACTION
213
VBp←VB
CA50Bp←CA50B
ΔCAFp←ΔCAFB+ΔCAC
END

FIG. 22


UPDATE LEARNED VALUE
301
OUTSIDE RANGE ?
NO
YES
302
READ VBp
303
READ ΔCAFp
304
CA50Fp←CA50Bp+ΔCAFp
305
CALCULATE VFp
306
ΔVFp←VFp−VBp
307a
CALCULATE r
307b
ΔVLd←ΔVFp*r
307c
RLf←ΔVFp*(1−r)
308
ΔCAFB←0
END

IGNITION TIMING CONTROL SYSTEM FOR A SPARK IGNITION TYPE INTERNAL COMBUSTION ENGINE

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to a spark ignition type internal combustion engine.

2. Related Art

Known in the art is an ignition timing control system of an internal combustion engine which uses the engine operating state as the basis to calculate a base ignition timing, calculates a feedback correction value for correcting the base ignition timing so that the level of knocking which is detected by the knocking sensor becomes a target value, and corrects the base ignition timing by the feedback correction value and a learned value (see Japanese Patent Publication No. 2011-256725A). It is not preferable in control for the feedback correction value to steadily deviate from the reference value. Therefore, in Japanese Patent Publication No. 2011-256725A, the steady difference of the feedback correction value with respect to the reference value is set as the learned value and the base ignition timing is corrected by the feedback correction value and learned value. As a result, the feedback correction value is kept from steadily deviating from the reference value.

In Japanese Patent Publication No. 2011-256725A, the learned value is set in relation to the engine operating state which is determined by the engine load and the engine speed.

SUMMARY OF THE INVENTION

However, the level of knocking depends on the state of combustion. The engine parameters which affect the state of combustion are not limited to the engine load and engine speed. For example, in an internal combustion engine in which the mechanical compression ratio can be changed, if the mechanical compression ratio is changed, the state of combustion is changed, so the level of knocking is changed. For this reason, in an internal combustion engine in which the mechanical compression ratio can be changed, even if using the learned value which is described in Japanese Patent Publication No. 2011-256725A to control the ignition timing, the level of knocking is liable to be unable to be maintained at the target level. In an internal combustion engine in which the closing timing of the intake valve can be changed as well, if the closing timing of the intake valve is changed, the level of knocking is changed, so a similar problem can occur.

If setting the learned value while considering the mechanical compression ratio or the closing timing of the intake valve, that is, the configuration of the internal combustion engine, this problem might be able to be solved. However, it would be extremely difficult and impractical to set the learned value in relation to all parameters which would be liable to affect the knocking.

According to the present invention, there is provided a spark ignition type internal combustion engine which uses the engine operating state as the basis to calculate a base ignition timing, calculates a feedback correction value for correcting the base ignition timing so that a level of knocking which is detected by a knock sensor becomes a target level, calculates the crank angle by which the combustion ratio becomes a predetermined set ratio when assuming combustion was performed by the base ignition timing as a "base crank angle" and an in-cylinder volume at the base crank angle as a "base in-cylinder volume", corrects the base in-cylinder volume by a learned value so as to calculate a corrected in-cylinder volume, calculates the crank angle by which the corrected in-cylinder volume is obtained as a "corrected crank angle", calculates a difference of the corrected crank angle from the base crank angle as a "crank angle correction value", and corrects the base ignition timing by the feedback correction value and crank angle correction value, the spark ignition type internal combustion engine calculating an in-cylinder volume at the crank angle by which the combustion ratio becomes a set ratio in the previous combustion as a "previous in-cylinder volume", calculating a difference of the previous in-cylinder volume from the base in-cylinder volume at the previous combustion, and updating the learned value based on the difference.

The present invention may be more fully understood from the description of the preferred embodiments according to the invention as set forth below, together with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

In the drawings:

FIG. **1** is an overall view of a spark ignition type internal combustion engine.

FIG. **2** is a disassembled perspective view of a variable compression ratio mechanism.

FIGS. **3**A and **3**B are side cross-sectional views of an internal combustion engine expressed schematically.

FIG. **4** is a view which shows a variable valve timing mechanism.

FIG. **5** is a view which shows amounts of lift of an intake valve and exhaust valve.

FIG. **6** is a view which shows a map of a base ignition timing SAB.

FIG. **7** is a time chart which explains a method of calculation of a feedback correction value.

FIG. **8** is a flow chart which shows a routine for calculation of a feedback correction value.

FIG. **9** is a graph which explains a relationship of a difference of ignition timing and a difference of crank angle by which a ratio of combustion becomes 50%.

FIG. **10** is a graph which shows a relationship of a crank angle correction value ΔCAC and an in-cylinder volume correction values ΔVC.

FIG. **11** is a view of a map of a base in-cylinder volume VB.

FIG. **12** is a view which shows a relationship among an in-cylinder volume, crank angle, and mechanical compression ratio ϵ.

FIG. **13** is a graph which explains a difference of amount of change of in-cylinder volume.

FIG. **14** is a time chart which shows a change of a learned value ΔVL.

FIG. **15** is a graph which explains a method of calculation of a learned value ΔVL.

FIGS. **16**A and **16**B are views which show maps of correction values Δvd**1** and Δvd**2**.

FIG. **17** is a flow chart of an ignition timing control routine.

FIG. **18** is a flow chart of a learned value calculation routine.

FIGS. **19**A and **19**B are views which show maps of correction values rd**1**, rd**2**.

FIG. **20** is a view which shows a map of a distribution coefficient "r".

FIG. **21** is a flow chart of an ignition timing control routine of another embodiment according to the present invention.

FIG. **22** is a flow chart of a learned value calculation routine of another embodiment according to the present invention.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

FIG. **1** is a side cross-sectional view of a spark ignition type internal combustion engine.

Referring to FIG. **1**, **1** indicates a crankcase, **2** a cylinder block, **3** a cylinder head, **4** a piston, **5** a combustion chamber, **6** a spark plug which is arranged at the top center of the combustion chamber **5**, **7** an intake valve, **8** an intake port, **9** an exhaust valve, and **10** an exhaust port. The intake port **8** is coupled through an intake runner **11** to a surge tank **12**. At each intake runner **11**, a fuel injector **13** is arranged for injecting fuel toward the corresponding intake port **8**. Note that a fuel injector **13** may also be arranged in each combustion chamber **5** instead of at each intake runner **11**.

The surge tank **12** is connected with an air cleaner **15** through an intake duct **14**. Inside the intake duct **14**, a throttle valve **17** which is driven by an actuator **16** and an intake air amount detector **18** which uses for example hot wires are arranged. On the other hand, the exhaust port **10** is connected through an exhaust manifold **19** to a catalyst converter **20** which has for example a three-way catalyst built into it. Inside the exhaust manifold **19**, an air-fuel ratio sensor **21** is arranged. Further, the cylinder block **2** is provided with a knock sensor **22** for detecting the level of knocking.

On the other hand, in the embodiment which is shown in FIG. **1**, at the connecting part of the crankcase **1** and the cylinder block **2**, a variable compression ratio mechanism A is provided which can change the relative positions of the crankcase **1** and the cylinder block **2** in the cylinder axial direction to as to change the mechanical compression ratio. Further, a variable valve timing mechanism B which can control the closing timing of the intake valve **7** is provided for controlling the intake air amount which is actually fed to the combustion chamber **5**. Note that the mechanical compression ratio is defined as the ratio of the in-cylinder volume when the piston is at compression bottom dead center to the in-cylinder volume when the piston is at compression top dead center.

An electronic control unit **30** is comprised of a digital computer. This is provided with components connected to each other through a bidirectional bus **31** such as a ROM (read only memory) **32**, RAM (random access memory) **33**, CPU (microprocessor) **34**, input port **35**, and output port **36**. An output signal of the intake air amount detector **18**, an output signal of the air-fuel ratio sensor **21**, and an output signal of the knock sensor **22** are input through respectively corresponding AD converters **37** to an input port **35**. Further, the accelerator pedal **40** is connected to a load sensor **41** which generates an output voltage which is proportional to the amount of depression L of the accelerator pedal **40**. The output voltage of the load sensor **41** is input through a corresponding AD converter **37** to the input port **35**. Furthermore, the input port **35** is connected to a crank angle sensor **42** which generates an output pulse each time a crankshaft rotates by for example 30°. On the other hand, the output port **36** is connected through a corresponding drive circuit **38** to each spark plug **6**, fuel injector **13**, an actuator **16** for driving the throttle valve, variable compression ratio mechanism A, and variable valve timing mechanism B.

FIG. **2** is a disassembled perspective view of the variable compression ratio mechanism A which is shown in FIG. **1**, while FIGS. **3**A and **3**B are a side cross-sectional view of the illustrated internal combustion engine. Referring to FIG. **2**, at the bottom of the two sides of the cylinder block **2**, a plurality of projecting parts **50** are formed spaced apart from each other. Inside these projecting parts **50**, cam insertion holes **51** of round cross-sectional shapes are formed. On the other hand, at the top surface of the crankcase **1**, a plurality of projecting parts **52** are formed which are spaced apart from each other and fit between the corresponding projecting parts **50**. Inside these projecting parts **52** as well, round cross-section cam insertion holes **53** are formed.

As shown in FIG. **2**, a pair of cam shafts **54** and **55** are provided. On these cam shafts **54** and **55**, at every other position, a circular cam **56** which is inserted into a cam insertion hole **53** is fastened. These circular cams **56** are coaxial with the axes of rotation of the cam shafts **54** and **55**. On the other hand, at the two sides of each circular cam **56**, as shown in FIGS. **3**A and **3**B, eccentric shafts **57** extend arranged eccentrically with respect to the axes of the cam shafts **54** and **55**. Other circular cams **58** are attached eccentrically on these eccentric shafts **57** in a rotatable manner. As shown in FIG. **2**, these circular cams **58** are arranged between the individual circular cams **56**. These circular cams **58** are rotatably inserted into the corresponding cam insertion holes **53**.

If making the circular cams **56** which are fastened on the cam shafts **54** and **55** are made to rotate from the state such as shown in FIG. **3**A in the mutually opposite directions as shown by the solid arrows in FIG. **3**A, the eccentric shafts **57** move toward the bottom center, so the circular cams **58** rotate in opposite directions from the circular cams **56** in the cam insertion holes **53** as shown by the broken line arrows in FIG. **3**A. When the eccentric shafts **57** move to the bottom centers as shown in FIG. **3**B, the centers of the circular cams **58** move below the eccentric shafts **57**.

As will be understood from a comparison of FIG. **3**A to FIG. **3**B, the relative positions of the crankcase **1** and the cylinder block **2** are determined by the distance between the centers of the circular cams **56** and the centers of the circular cams **58**. The larger the distance between the centers of the circular cams **56** and the centers of the circular cams **58**, the more the cylinder block **2** separates from the crankcase **1**. **2**. If the cylinder block **2** moves away from the crankcase **1**, the volume of the combustion chamber **5** when the piston **4** is positioned at compression top dead center increases. Therefore, by rotating the cam shafts **54** and **55**, the volume of the combustion chamber **5** when the piston **4** is positioned at compression top dead center can be changed. As shown in FIG. **2**, to make the cam shafts **54** and **55** rotate in opposite directions, the shaft **60** of a drive motor **59** is formed with a pair of worms **61** and **62** with opposite spiral directions. The worm gears **63** and **64** which engage with these worms **61** and **62** are fastened to the ends of the cam shafts **54** and **55**. In this embodiment, by operating the drive motor **59**, the volume of the combustion chamber **5** when the piston **4** is positioned at compression top dead center can be changed over a wide range. Note that the variable compression ratio mechanism A which is shown from FIG. **1** to FIG. **3**B is one example. Any type of variable compression ratio mechanism can be used.

On the other hand, FIG. **4** shows a variable valve timing mechanism B which is attached to an end of a cam shaft **70** for driving the intake valve **7** in FIG. **1**. Referring to FIG. **4**, this variable valve timing mechanism B is provided with a timing pulley **71** which is made to rotate in the arrow direction by a crankshaft of the engine through a timing belt, a cylindrical housing **72** which rotates together with the timing pulley **71**, a shaft **73** which can rotate relative together with the cam shaft **70** for driving the intake valve and can rotate relative to the cylindrical housing **72**, a plurality of partition walls **74** which extend from the inner circumference of the cylindrical housing **72** to the outer circumference of the shaft **73**, and vanes **75** which extend between the partition walls **74** from the outer circumference of the shaft **73** to the inner circumference of the cylindrical housing **72**.

The feed of working oil to the hydraulic chambers **76** and **77** is controlled by a working oil feed control valve **78**. This working oil feed control valve **78** is provided with hydraulic ports **79** and **80** which are respectively connected to the hydraulic chambers **76** and **77**, a feed port **82** of working oil which is discharged from the hydraulic pump **81**, a pair of drain ports **83** and **84**, and a spool valve **85** which controls the connections/disconnections of the ports **79**, **80**, **82**, **83**, and **84**.

When advancing the phase of the cams of the cam shaft **70** for driving the intake valve, the spool valve **85** is made to move to the right in FIG. **4**, working oil which is fed from the feed port **82** is fed through a hydraulic port **79** to the hydraulic chamber **76** for advance, and the working oil inside the hydraulic chamber **77** for retard is drained from a drain port **84**. At this time, the shaft **73** is made to rotate relative to the cylindrical housing **72** in the arrow direction.

As opposed to this, when retarding the phase of the cam of the cam shaft **70** for driving the intake valve, the spool valve **85** is made to move to the left in FIG. **4**, working oil which is fed from the feed port **82** is fed through a hydraulic port **80** to the hydraulic chamber **77** for retard, and the working oil inside the hydraulic chamber **76** for advance is drained from a drain port **83**. At this time, the shaft **73** is made to rotate relative to the cylindrical housing **72** in a direction opposite to the arrow.

When the shaft **73** is made to rotate relative to the cylindrical housing **72**, if the spool valve **85** is returned to the neutral position which is shown in FIG. **4**, the relative rotation operation of the shaft **73** is made to stop and the shaft **73** is held at the relative rotation position at that time. Therefore, the variable valve timing mechanism B can be used to make the phase of the cams of the cam shaft **70** for driving the intake valve advance and be retarded by a desired amount.

In FIG. **5**, the solid line shows when the variable valve timing mechanism B makes the phase of the cams of the cam shaft **70** for driving the intake valve advance the most, while the broken line shows when it makes the phase of the cams of the cam shaft **70** for driving the intake valve be retarded the most. Therefore, the opening interval of the intake valve **7** can be freely set between the range shown by the solid line and the range shown by the broken line in FIG. **5**. Therefore, the closing timing of the intake valve **7** also can be set to any crank angle in the range shown by the arrow C in FIG. **5**.

The variable valve timing mechanism B which is shown in FIG. **1** and FIG. **4** shows one example. For example, it is possible to use a variable valve timing mechanism which can change only the closing timing of the intake valve while maintaining the opening timing of the intake valve constant and various other types of variable valve timing mechanisms.

In the embodiment according to the present invention, for example, the following cycles are performed. That is, when the engine load is higher than the first set load, a cycle is performed where the intake air amount is controlled by maintaining the mechanical compression ratio at a predetermined compression ratio or less and maintaining the closing timing of the intake valve at the side near suction bottom dead center while controlling the throttle opening degree. When the engine load is lower than a first set load and is higher than a second set load which is set lower than the first set load, a cycle is performed where the intake air amount is controlled by maintaining the mechanical compression ratio at a predetermined compression ratio or more, for example, at the maximum mechanical compression ratio, and maintaining the throttle valve wide open while controlling the closing timing of the intake valve. When the engine load is lower than the second set load, a cycle is performed where the intake air amount is controlled by maintaining the mechanical compression ratio at a predetermined compression ratio or more and maintaining the closing timing of the intake valve at the side far from suction bottom dead center while controlling the throttle opening degree.

Now, in the embodiment according to the present invention, the ignition timing SAF is calculated from the following formula (1):

$$SAF=SAB+\Delta CAFB+\Delta CAC \tag{1}$$

In formula (1), SAB indicates the base ignition timing, ΔCAFB the feedback correction value, and ΔCAC the crank angle correction value.

The base ignition timing SAB is the ignition timing which is required for making the level of knocking the target level when the engine state is the reference state and is stored as a function of the engine operating state, for example, the engine load L and engine speed Ne, in the form of the map which is shown in FIG. **6** in advance in the ROM **32**. Note that the "reference state" means the state where the parameters which affect the level of knocking, such as the intake temperature, engine cooling water temperature, amount of deposits which form on the inside wall of the combustion chamber, and fuel properties such as the octane value, are at their corresponding reference values.

The feedback correction value ΔCAFB is for correcting the base ignition timing SAB when the level of knocking which is detected by the knock sensor **22** becomes the target level. It is set to the reference value zero when correction is not required.

That is, as shown in FIG. **7**, if, at the time ta**1**, the level of knocking KLV becomes larger than the upper limit value KLVU of the target level TKLV, the feedback correction value ΔCAFB is increased. That is, the ignition timing SAF is retarded. Next, if, at the time ta**2**, the level of knocking KLV becomes the target level TKLV, the feedback correction value ΔCAFB is held. Next, if, at the time ta**3**, the level of knocking KLV becomes smaller than the lower limit value KLVL of the target level TKLV, the feedback correction value ΔCAFB is reduced. That is, the ignition timing SAF is advanced. Next, if, at the time ta**4**, the level of knocking KLV becomes the target level TKLV, the feedback correction value ΔCAFB is held.

FIG. **8** shows a routine for calculation of the feedback correction value ΔCAFB. This routine is executed by interruption every certain time period.

Referring to FIG. **8**, at step **101**, it is judged if the level of knocking KLV is larger than the upper limit value KLVU. When KLV>KLVU, next, the routine proceeds to step **102** where the feedback correction value ΔCAFB is increased by a constant value dFB. As opposed to this, when KLV≤KLVU, next, the routine proceeds to step **103** where it is judged if the level of knocking KLV is smaller than the lower limit value KLVL. When KLV<KLVL, next the routine proceeds to step **104** where the feedback correction value ΔCAFB is reduced by a certain value dFB. As opposed to this, when KLV≥KLVL, the processing cycle is ended. That is, the feedback correction value ΔCAFB is held.

On the other hand, the crank angle correction value ΔCAC is a conversion of the later explained in-cylinder volume correction value ΔVC to the crank angle. It is set to the reference value zero when correction is not required.

In this regard, FIG. **9** schematically shows the combustion period X when combustion is performed by the ignition timing SAX. At this time, the crank angle at which the combustion ratio becomes a predetermined set ratio is shown by CA**50**X. In this embodiment according to the present invention, the set ratio is made 50%. Further, the combustion period Y when combustion is performed by the ignition timing SAY is schematically shown. At this time, the crank angle where the combustion ratio becomes 50% is shown by CA**50**Y. In the example which is shown in FIG. **9**, at the combustion periods X, Y, the ignition timings differ by ΔSA (=SAY−SAX) and the crank angles where the combustion ratios become 50% differ by ΔCA**50** (=CA**50**Y−CA**50**X).

In this embodiment according to the present invention, it is assumed that the difference ΔSA of the ignition timing and the difference ΔCA**50** of the crank angle where the combustion ratio becomes 50% are equal to each other. Therefore, for example if the ignition timing is retarded by the constant crank angle ΔCA, the crank angle where the combustion ratio becomes 50% is also retarded by the constant crank angle ΔCA. In other words, to retard the crank angle where the combustion ratio becomes 50% by a constant crank angle ΔCA, it is sufficient to retard the ignition timing by ΔCA.

Note that if expressing the in-cylinder pressure and in-cylinder volume at any crank angle by P and V, expressing the in-cylinder pressure and in-cylinder volume at the time of start of combustion by P**0** and V**0**, expressing the in-cylinder pressure and in-cylinder volume at the time of end of combustion by Pf and Vf, and expressing the specific heat ratio as κ, the combustion ratio (%) is expressed by the following formula (2).

$$\text{Combustion ratio (\%)}=(P\cdot V^{\kappa}-P0\cdot V0^{\kappa})/(Pf\cdot Vf^{\kappa}-P0\cdot V0^{\kappa})\cdot 100 \quad (2)$$

Next, referring to FIG. **10**, control for correcting the ignition timing in an embodiment according to the present invention will be explained.

In this embodiment according to the present invention, first, if referring to the crank angle where the combustion ratio becomes 50% when assuming that combustion is performed by the base ignition timing SAB, that is, when assuming no correction is performed, as the "base crank angle CA**50**B", the in-cylinder volume at the base crank angle CA**50**B is calculated as the "base in-cylinder volume VB". The base in-cylinder volume VB is stored as a function of the engine operating state, for example, the engine load L and engine speed Ne, in the form of the map which is shown in FIG. **11** in advance in the ROM **32**.

Next, the base in-cylinder volume VB is corrected to calculate the corrected in-cylinder volume VC.

On the other hand, the base crank angle CA**50**B is calculated from the base in-cylinder volume VB and mechanical compression ratio ϵ. Further, the crank angle by which the corrected in-cylinder volume VC is obtained is calculated as the "corrected crank angle CA**50**C" from the corrected in-cylinder volume VC and mechanical compression ratio ϵ. In this embodiment according to the present invention, the relationship among the crank angle CA, in-cylinder volume V, and mechanical compression ratio ϵ are stored in the form of the map which is shown in FIG. **12** in advance in the ROM **32**.

Next, the difference ΔCAC of the corrected crank angle CA**50**C from the base crank angle CA**50**B is calculated (ΔCAC=CA**50**C−CA**50**B). This difference ΔCAC is the crank angle which is necessary for changing the base in-cylinder volume VB to the corrected in-cylinder volume VC and is set to the above-mentioned crank angle correction value ΔCAC.

That is, to correct the base in-cylinder volume VB to the corrected in-cylinder volume VC, the base crank angle CA**50**B is corrected by the crank angle correction value ΔCAC. To correct the base crank angle CA**50**B by the crank angle correction value ΔCAC, the base ignition timing SAB is corrected by the crank angle correction value ΔCAC.

In this way, in this embodiment according to the present invention, the base in-cylinder volume VB is corrected. This is done for the following reason. That is, according to the inventors etc., it is learned that the level of knocking is strongly correlated with the in-cylinder volume during the combustion period. Therefore, to accurately control the level of knocking, it is necessary to accurately control the in-cylinder volume in the combustion period. On the other hand, as shown in FIG. **13**, the amount of change of the in-cylinder volume which is obtained by changing the crank angle by the constant value dCA differs according to the crank angle before the change. That is, when the crank angle is close to compression top dead center TDC, that is, when the crank angle is small, the in-cylinder volume changes by a comparatively small amount dVS, while when the crank angle is far from compression top dead center TDC, that is, when the crank angle is large, the in-cylinder volume changes by a relatively large amount dVL. For this reason, if correcting the base ignition timing VB simply by the crank angle dCA, when the base ignition timing SAB is small, the amount of change of the in-cylinder volume becomes small, while when the base ignition timing SAB is large, the amount of change of the in-cylinder volume ends up becoming large. As a result, it becomes difficult to maintain the in-cylinder volume at the combustion period at an in-cylinder volume which maintains the level of knocking at the target level.

Therefore, in this embodiment according to the present invention, the corrected in-cylinder volume VC is calculated, the crank angle which is necessary for changing the base ignition timing SAB to the corrected in-cylinder volume VC, is calculated as the "crank angle correction value ΔCAC", and the crank angle correction value ΔCAC is used to correct the base ignition timing SAB.

In the embodiment according to the present invention, the corrected in-cylinder volume VC is calculated in the following way. That is, first, the in-cylinder volume correction value ΔVC is calculated. Next, the corrected in-cylinder volume VC is calculated from the following formula (3).

$$VC=VB+\Delta VC \quad (3)$$

In the embodiment according to the present invention, the in-cylinder volume correction value ΔVC is calculated from the following formula (4).

$$\Delta VC=\Delta VL+\Delta VD \quad (4)$$

In formula (4), ΔVL indicates the learned value, while ΔVD indicates a disturbance correction value.

The learned value ΔVL is for compensating for disturbances which cannot be individually judged among the disturbances causing the engine state to deviate from the reference state, for example, the amount of deposits which form on the wall surface inside of a combustion chamber and the properties of the fuel. When correction is not required, it is made the reference value zero. On the other hand, the disturbance correction value ΔVD is for compensating for disturbances which can be detected by sensors etc. among the disturbances causing the engine state to deviate from the reference state, for example, the intake temperature and the engine cooling water temperature. When correction is not required, it is made the reference value zero.

The learned value ΔVL is updated so that the feedback correction value ΔCAFB does not greatly deviate from the reference value. That is, as shown in FIG. **14**, if, at the time tb**1**, the feedback correction value ΔCAFB becomes larger than the upper limit value ARU of the allowable range AR, the learned value ΔVL is increased in a step manner and the feedback correction value ΔCAFB is returned to the reference value in a step manner. On the other hand, if, at the time tb**2**, the feedback correction value ΔCAFB becomes smaller than the lower limit value ARL of the allowable range AR, the learned value ΔVL is reduced in a step manner and the feedback correction value ΔCAFB is returned to the reference value in a step manner. If doing this, the feedback correction value ΔCAFB is kept from greatly deviating from the reference value and the controllability is improved.

Next, referring to FIG. **15**, the method of calculation of the learned value ΔVL will be explained. If referring to FIG. **15**, the base in-cylinder volume VBp at the previous combustion, the base crank angle CA**50**Bp at the previous combustion, the crank angle CA**50**Fp where the combustion ratio became 50% in the previous combustion, and the in-cylinder volume where the combustion ratio becomes 50% at the crank angle CA**50**Fp at the previous combustion, that is, the previous in-cylinder volume VFp, are shown. Further, in FIG. **15**, ΔCAFp shows the difference of the crank angle CA**50**Fp where the combustion ratio became 50% in the previous combustion from the base crank angle CA**50**Bp at the previous combustion (ΔCAFp=CA**50**Fp−CA**50**Bp), while ΔVFp shows the difference of the previous in-cylinder volume VFp from the base in-cylinder volume VBp at the previous combustion (ΔVFp=VFp−VBp).

In the previous combustion, the base ignition timing SAB is corrected by the feedback correction value ΔCAFB and crank angle correction value ΔCAC. Due to this, the base crank angle CA**50**Bp is corrected by the crank angle difference ΔCAFp. If converting the crank angle difference ΔCAFp to the difference in in-cylinder volume, it becomes the in-cylinder volume difference ΔVFp. That is, at the previous combustion, to maintain the level of knocking at the target level, it was necessary to correct the base in-cylinder volume VBp by the in-cylinder volume difference ΔVFp.

Therefore, in this embodiment according to the present invention, this in-cylinder volume difference ΔVFp is set to the learned value ΔVL. As a result, it is possible to make steady corrections to the base in-cylinder volume VB by the learned value ΔVL.

The in-cylinder volume difference ΔVFp is calculated as follows. That is, the crank angle difference ΔCAFp at the previous combustion is stored as the total of the feedback correction value ΔCAFB and crank angle correction value ΔCAC at the previous combustion. The crank angle difference ΔCAFp at the previous combustion is added to the base crank angle CA**50**Bp in the previous combustion to calculate the crank angle CA**50**Fp where by combustion ratio became 50% at the previous combustion (CA**50**Fp=CA**50**Bp+ΔCAFp). Next, the in-cylinder volume VFp at the crank angle CA**50**Fp is calculated by using the map of FIG. **12**. Next, the in-cylinder volume difference ΔVFp is calculated (ΔVFp=VFp−VBp).

On the other hand, the disturbance correction value ΔVD is for example calculated from the following formula (5).

$$\Delta VD=\Delta vd1+\Delta vd2 \tag{5}$$

The correction value Δvd**1** is a correction value which compensates for the deviation of the intake temperature from the reference intake temperature which forms the above-mentioned reference state, while the correction value Δvd**2** is a correction value which compensates for the deviation of the engine cooling water temperature THW from the reference water temperature which constitutes the above-mentioned reference state. These correction values Δvd**1** and Δvd**2** are stored as values which are converted to differences of the in-cylinder volume in the form of the maps which are shown in FIGS. **16**A and **16**B in advance in the ROM **32**. Note that the intake temperature Ta is detected by an intake temperature sensor which is attached to the intake duct **14**, while the engine cooling water temperature THW is detected by a water temperature sensor which is attached to the cylinder block **2**. In another embodiment, other parameters which form the reference state, for example, the engine lubricant temperature, may also be considered.

FIG. **17** shows the routine of control of the ignition timing in an embodiment according to the present invention. This routine is executed by interruption every certain time period.

Referring to FIG. **17**, at step **201**, the base ignition timing SAB is calculated by using the map of FIG. **6**. At the next step **202**, the routine for calculation of the feedback correction value ΔCAFB is performed. This routine is shown in FIG. **8**. At the next step **203**, the base in-cylinder volume VB is calculated by using the map of FIG. **11**. At the next step **204**, the learned value ΔVL is read. The learned value ΔVL is updated by the routine which is shown in FIG. **18**. At the next step **205**, the correction values Δvd**1**, Δvd**2** are calculated by using the maps of FIGS. **16**A and **16**B and disturbance correction value ΔVD is calculated (ΔVD=Δvd**1**+Δvd**2**). At the next step **206**, the in-cylinder volume correction values ΔVC is calculated (ΔVC=ΔVL+ΔVD). At the next step **207**, the corrected in-cylinder volume VC is calculated (VC=VB+ΔVC). At the next step **208**, the base crank angle CA**50**B is calculated by using the map of FIG. **12**. At the next step **209**, the corrected crank angle CA**50**C is calculated by using the map of FIG. **12**. At the next step **210**, the crank angle correction value ΔCAC is calculated (ΔCAC=CA**50**C−CA**50**B). At the next step **211**, the ignition timing SAF is calculated (SAF=SAB+ΔCAC+ΔCAFB). At the next step **212**, the ignition action is performed by the ignition timing SAF. At the next step **213**, the base in-cylinder volume VB and base crank angle CA**50**B in the present routine are stored as the base in-cylinder volume VBp and base crank angle CA**50**B at the previous combustion. Further, the total of the correction values in the current routine is stored as the crank angle difference ΔCAFp at the previous combustion (ΔCAFp=ΔCAFB+ΔCAC).

FIG. **18** shows the routine for updating the learned value ΔVL in an embodiment according to the present invention. This routine is executed by interruption every certain time period.

Referring to FIG. **18**, at step **301**, it is judged if the feedback correction value ΔCAFB is inside the allowable range AR. When the feedback correction value ΔCAFB is inside the allowable range AR, the processing cycle is ended. That is, the feedback correction value ΔCAFB is held. As opposed to this, when the feedback correction value ΔCAFB is outside the allowable range AR, next the routine proceeds to step **302** where the base in-cylinder volume VBp at the previous combustion is read. At the next step **303**, the crank angle difference ΔCAFp at the previous combustion is read. At the next step **304**, the crank angle CA**50**Fp at the previous combustion is calculated (CA**50**Fp=CA**50**Bp+ΔCAFp). At the next step **305**, the in-cylinder volume VFp at the crank angle CA**50**Fp is calculated from the map of FIG. **12**. At the next step **306**, the in-cylinder volume difference ΔVFp is calculated (ΔVFp=VFp−VBp). At the next step **307**, the in-cylinder volume difference ΔVFp is set to the learned value ΔVL. At the next step **308**, the feedback correction value ΔCAFB is returned to the reference value zero.

Therefore, in the embodiment according to the present invention, the base ignition timing SAB is calculated, the feedback correction value ΔCAFB is calculated, the base crank angle CA**50**B and the base in-cylinder volume VB are calculated, and the base in-cylinder volume VB is corrected by the learned value ΔVL whereby the corrected in-cylinder volume VC is calculated, the corrected crank angle CA**50**C is calculated, the crank angle correction value ΔCAC is calculated, and the base ignition timing SAB is corrected by the feedback correction value ΔCAFB and crank angle correction value ΔCAC. Further, the previous in-cylinder volume VFp is calculated, the in-cylinder volume difference ΔVFp is calculated, and the in-cylinder volume difference ΔVFp is used as the basis to update the learned value ΔVL.

In this regard, the previous in-cylinder volume VFp which was explained with reference to FIG. **15** can be expressed using the in-cylinder volume difference ΔVFp as shown in the following formula (6).

$$VFp=VBp+\Delta VFp \tag{6}$$

In the above-mentioned embodiment, the in-cylinder volume difference ΔVFp is set to the learned value ΔVL. In this case, it may be considered that the in-cylinder volume difference ΔVFp has been replaced with a learned value ΔVL in the form of the in-cylinder volume difference.

As opposed to this, if using the rate of change RL of the in-cylinder volume (=ΔVFp/VBp), the previous in-cylinder volume VFp can be expressed as shown in the following formula (7).

$$VFp=VBp\cdot(1+RL) \tag{7}$$

On the other hand, the above-mentioned disturbance correction value ΔVD can be considered to compensate for disturbance which can be detected by a sensor etc. in the form of an in-cylinder volume difference.

However, disturbance which can be detected by a sensor etc. can also be compensated by a disturbance correction value in the form of the rate of change of the in-cylinder volume. If expressing the disturbance correction value in this case by RD, the disturbance correction value RD is calculated from, for example, the following formula (8).

$$RD=rd1\cdot rd2 \tag{8}$$

The correction value rd**1** is a correction value which compensates for the deviation of the intake temperature from the reference intake temperature which forms the reference state, while the correction values rd**2** is a correction value which compensates for the deviation of the engine cooling water temperature THW from the above reference water temperature which forms the reference state. These correction values rd**1** and r**2** are stored as values which are converted to rates of change of the in-cylinder volume in the form of the maps which are shown in FIGS. **19**A and **19**B in advance in the ROM **32**.

This being so, if setting the rate of change of the in-cylinder volume RL to the learned value and using the disturbance correction value RD, the corrected in-cylinder volume VC can be calculated from the following formula (9).

$$VC=VB\cdot(1+RL)\cdot(1+RD) \tag{9}$$

In this case, it may be considered that the in-cylinder volume difference ΔVFp has been replaced with a learned value RL in the form of the rate of change of the in-cylinder volume. However, the learned value RL in this case is not based on the in-cylinder volume difference but is updated based on the rate of change of the in-cylinder volume. As opposed to this, in the embodiment which is shown from FIG. **1** to FIG. **18**, the learned value ΔVL is not based on the rate of change of the in-cylinder volume, but is updated based on the in-cylinder volume difference.

Next, another embodiment according to the present invention will be explained. In this other embodiment according to the present invention, the corrected in-cylinder volume VC is calculated from the following formula (10).

$$VC=(VB+\Delta VLd)\cdot(1+RLf)\cdot(1+RD) \tag{10}$$

ΔVLd indicates a learned value in the form of the in-cylinder volume difference, while RLf indicates a learned value in the form of the rate of change of the in-cylinder volume. These are calculated by the following formulas (11) and (12) by using the distribution coefficient "r" ($0 \le r \le 1$).

$$\Delta VLd=\Delta VFp\cdot r \tag{11}$$

$$RLf=\Delta VFp\cdot(1-r) \tag{12}$$

In this case, it may be considered that part of the in-cylinder volume difference ΔVFp which is shown in FIG. **15** has been replaced with a learned value ΔVLd in the form of the in-cylinder volume difference, while the remainder of the in-cylinder volume difference ΔVFp has been replaced with a learned value RLf in the form of the rate of change of the in-cylinder volume.

According to the inventors etc., it was confirmed that to compensate for the amount of deposits which form on the wall surface inside of a combustion chamber, it is preferable to use a learned value ΔVLd in the form of the in-cylinder volume difference, while to compensate for the fuel properties such as the octane value, it is preferable to use a learned value RLf in the form of the rate of change of the in-cylinder volume. This is because if the amount of deposits which form on the wall surface inside of a combustion chamber changes, the in-cylinder volume will change by the amount of change of the deposits and the amount of change of the deposits is not dependent on the base in-cylinder volume. On the other hand, if the fuel properties change, the in-cylinder volume itself will not change, however the in-cylinder temperature and the in-cylinder pressure which affect the level of knocking will change. The amounts of change of the in-cylinder temperature and the in-cylinder pressure depend on the base in-cylinder volume. Therefore, in another embodiment according to the present invention, the two learned values ΔVLd, RLf are used. As a result, the level of knocking can be reliably maintained at the target level.

The distribution coefficient "r" is stored as a function of the engine operating state, for example, the engine load L and engine speed Ne, in the form of a map which is shown in FIG. **20** in advance in the ROM **32**.

FIG. **21** shows the routine for control of the ignition timing in another embodiment according to the present invention. This routine is executed by interruption every certain time period.

Referring to FIG. **21**, at step **201**, the base ignition timing SAB is calculated by using the map of FIG. **6**. At the next step **202**, the routine for calculation of the feedback correction value ΔCAFB is executed. This routine is shown in FIG. **8**. At the next step **203**, the base in-cylinder volume VB is calculated by using the map of FIG. **11**. At the next step **204***a*, the learned values ΔVLd, RLf are read. The learned values ΔVLd, RLf are updated by the routine which is shown in FIG. **22**. At the next step **205***a*, the correction values rd**1**, rd**2** are calculated by using the maps of FIGS. **19**A and **19**B and the disturbance correction value RD is calculated (RD=rd**1**·rd**2**). At the next step **207***a*, the corrected in-cylinder volume VC is calculated (VC=(VB+ΔVLd)·(1+RLf)·(1+RD)). At the next step **208**, the base crank angle CA**50**B is calculated by using the map of FIG. **12**. At the next step **209**, the corrected crank angle CA**50**C is calculated by using the map of FIG. **12**. At the next step **210**, the crank angle correction value ΔCAC is calculated (ΔCAC=CA**50**C−CA**50**B). At the next step **211**, the ignition timing SAF is calculated (SAF=SAB+ΔCAC+ΔCAFB). At the next step **212**, the ignition action is performed by the ignition timing SAF. At the next step **213**, the base in-cylinder volume VB and base crank angle CA**50**B in the current routine are stored as the base in-cylinder volume VBp and base crank angle CA**50**B in the previous combustion. Further, the total of the correction values at the current routine is stored as the crank angle difference ΔCAFp at the previous combustion (ΔCAFp=ΔCAFB+ΔCAC).

FIG. **22** shows the routine for updating the learned value ΔVL of another embodiment according to the present invention. This routine is executed by interruption every certain time period.

Referring to FIG. **22**, at step **301**, it is judged if the feedback correction value ΔCAFB is in the allowable range AR. When the feedback correction value ΔCAFB is in the allowable range AR, the processing cycle is ended. That is, the feedback correction value ΔCAFB is maintained. As opposed to this, when the feedback correction value ΔCAFB is outside of the allowable range AR, next the routine proceeds to step **302** where the base in-cylinder volume VBp at the previous combustion is read. At the next step **303**, the crank angle difference ΔCAFp at the previous combustion is read. At the next step **304**, the crank angle CA**50**Fp at the previous combustion is calculated (CA**50**Fp=CA**50**Bp+ΔCAFp). At the next step **305**, the in-cylinder volume VFp at the crank angle CA**50**Fp is calculated from the map of FIG. **12**. At the next step **306**, the in-cylinder volume difference ΔVFp is calculated (ΔVFp=VFp−VBp). At the next step **307***a*, the distribution coefficient "r" is calculated by using the map of FIG. **20**. At the next step **307***b*, the learned value ΔVLd in the form of the in-cylinder volume difference is calculated (ΔVLd=ΔVFp·r). At the next step **307***c*, the learned value RLf in the form of the rate of change of the in-cylinder volume is calculated (RLf=ΔVFp·(1−r)). At the next step **308**, the feedback correction value ΔCAFB is returned to the reference value zero.

In this way, in another embodiment according to the present invention, the learned value is updated based on the in-cylinder volume difference and rate of change of the in-cylinder volume.

The present invention can reliably maintain the level of knocking at a target level regardless of the configuration of an internal combustion engine.

While the invention has been described by reference to specific embodiments chosen for purposes of illustration, it should be apparent that numerous modifications could be made thereto, by those skilled in the art, without departing from the basic concept and scope of the invention.

This application claims the benefit of JP Application No. 2013-060523, the entire disclosure of which is incorporated by reference herein.

The invention claimed is:

1. An ignition timing control system comprising:

a spark ignition type internal combustion engine;

a knock sensor that detects a level of knocking of the spark ignition type internal combustion engine; and

an electronic control unit operatively connected to the spark ignition type internal combustion engine, the electronic control unit configured to:

calculate a base ignition timing based on an engine operating state of the spark ignition type internal combustion engine,

calculate a feedback correction value that corrects the base ignition timing so that the detected level of knocking by the knock sensor becomes a target level,

calculate a crank angle by which a combustion ratio becomes a predetermined set ratio when assuming combustion was performed by the base ignition timing as a base crank angle and an in-cylinder volume at the base crank angle as a base in-cylinder volume,

correct the base in-cylinder volume by a learned value so as to calculate a corrected in-cylinder volume,

calculate the crank angle by which the corrected in-cylinder volume is obtained as a corrected crank angle,

calculate a difference of the corrected crank angle from the base crank angle as a crank angle correction value,

correct the base ignition timing by the feedback correction value and crank angle correction value,

calculate an in-cylinder volume at the crank angle by which the combustion ratio becomes a set ratio in the previous combustion as a previous in-cylinder volume,

calculate a difference of the previous in-cylinder volume from the base in-cylinder volume at the previous combustion,

update the learned value based on the difference of the previous in-cylinder volume from the base in-cylinder volume at the previous combustion, and

control the spark ignition type internal combustion engine based on the updated learned value.

2. The ignition timing control system according to claim **1**, wherein the electronic control unit is configured to update the learned value based on the difference of the previous in-cylinder volume from the base in-cylinder volume at the previous combustion and a rate of change of the previous in-cylinder volume from the base in-cylinder volume at the previous combustion.

3. The ignition timing control system according to claim **1**, wherein the electronic control unit is configured to maintain the learned value when the feedback correction value is within an allowable range and update the learned value when the feedback correction value is outside the allowable range.

4. The ignition timing control system according to claim **1**, wherein the set ratio is 50%.

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